Flare Note **63**
Hans Jostlein 12/30/2004

Friction of Steel Wire on Pins, Warm and Cold

Abstract

Very large liquid Argon time projection chambers may use detection planes composed of long wires, stretched by weights. Non--vertical wires have to make a turn, e.g. around a pin, to the vertical for weight attachment.
We have measured the static friction of 0.150 mm diameter stainless steel wire on steel, brass, and ceramic pins at room temperature and in liquid nitrogen.
Friction coefficients range from 0.21 for warm glazed ceramic to 0.35 for warm and cold steel pins (dynamic friction is not much lower).
These friction coefficients meet the needs of the experiment.
However, we conclude that pulleys or lower fiction materials will be desirable for improved uniformity of wire tension.

Introduction

The FLARE group is building a small Liquid Argon prototype TPC and is also doing conceptual engineering studies for a large neutrino TPC
The large TPC would have about 25 kton of Argon.
The drift distance would be 3 m, and the induction and collection wire planes would be composed of stretched stainless alloy wires of 0.15 mm diameter and up to 30 m length. They would be stretched with 1.3 kg weights.
The close vicinity of wires, combined with the large weight dimensions, may force a design where the wire ends are separated from one another by a set of pins or pulleys. Non-vertical wires always need a deflection element.
Any friction force in those pins or pulleys will alter the pre-load of the wire tension and cause irregularities in their spacing.
The relevant friction here is the static friction. As the temperature drops on cool-down, wires will be initially at rest and will relieve the tension built up from thermal contraction when that tension force overcomes the static friction force. Each wire will come to rest anywhere, unpredictably, within the weight tension plus or minus the static friction force.
We have measured the friction coefficient of the 0.15 mm wire under 1,300 g tension against pins made of steel, brass, and ceramic. Steel and ceramic were measured both warm and cold. The device involved 5 ball bearing pulleys, and we have “friction coefficients” for 5 warm pulleys (taken together) as well as 4 warm and 1 cold pulley.

The Test Apparatus

As seen in Fig. 1, the apparatus uses two weights, each close to 1300g, and 5 pulleys, that take the tensioned wire into and out of a liquid nitrogen bath.

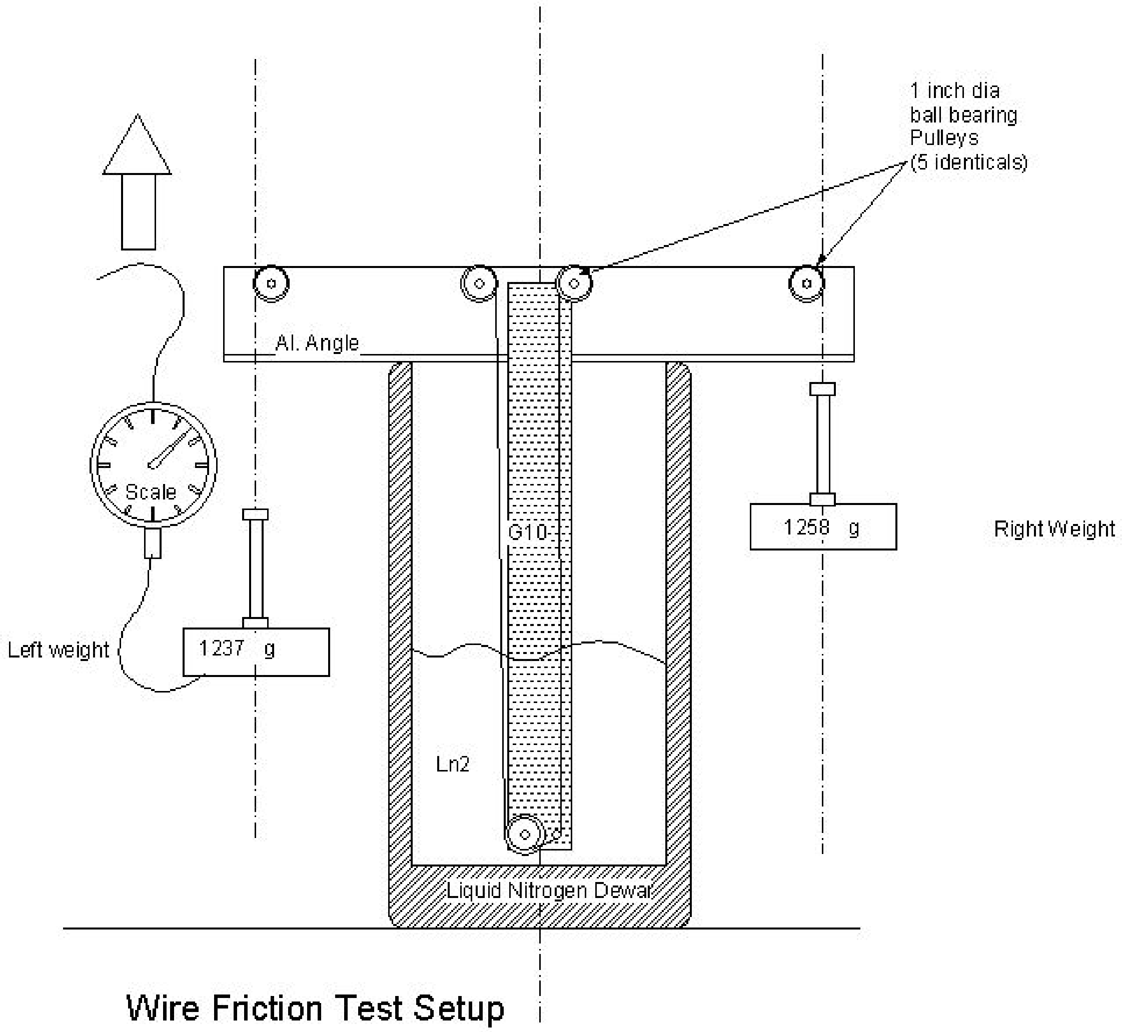

Figure 1

A hand-held spring scale is used to lift up the left or right weight until the weights and the wire move. The highest reading on the spring scale is the force it took to overcome the static friction.

Note, by the way, that the dynamic friction, i.e. the force reading during motion, was almost as high as the static friction, maybe 80% or 90% as large.

Analysis Details

We write the friction force as being proportional to the contact force times a friction coefficient “rho”.
For a weight on a plane the variables are obvious.
For a ball bearing we assumed that the total shaft force was the right contact force to use.
For a wire wrapped around a pin one needs to integrate the contact force (normal to the surface) over the contact angle.
The total contact force is the tension force times the contact angle.
Figure 2 shows the two contact angles used here. They differ slightly due to the different pin diameters used.
Academic note: if the friction force is a good fraction of the tension, we have a “capstan” situation where the force changes exponentially with angle. The average tension would then be calculated by averaging over the logarithm of the tension. I have used a linear approximation here.
The measurement was repeated about a dozen times, alternating left and right, to get the mean and the variation of the required force. The average was taken of left and right numbers, and it was further corrected for the scale offset and the reduction in tension due to the up-force exerted by the spring scale. The small friction force contributed by the ball bearing pulleys was also subtracted.
In all cases the pin and wire were degreased with alcohol prior to the tests.
In the case of brass, there is evidence that some brass rubbed off onto the wire.
The ceramic pin was measured after the brass, and exhibited a smear pattern where the wire ran. This effect could have been removed by replacing the wire after each measurement. However the results will show that it doesn’t matter.

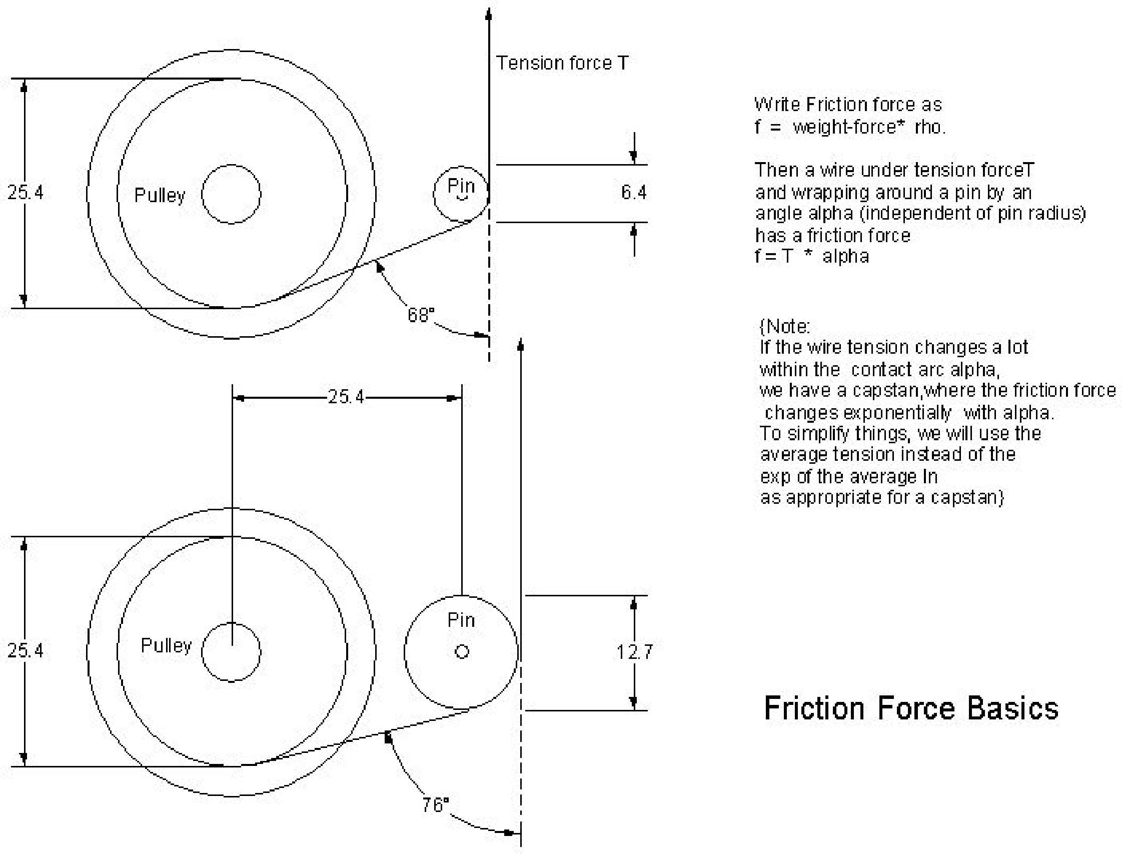

Figure 2: Friction Force Geometry

Friction Coefficient for 0.150 mm steel wire on pins, warm and cold

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Bolt weight	65 g				
Scale offset (Tare)	25 g				
Left weight	1237 g		Right Weight	1258 g	
with bolt	1302 g		with bolt	1323 g	
Average Tension				1290.5 g	
Pi	3.141592654				
Wire deflection for 1/4 inch pin		68 deg		1.187 rad	
Wire deflection for 1/2 inch pin		76 deg		1.326 rad	
Wheel total axle loading		6231 g			
Express as contact angle		4.828 radian			

	5 Ball Bearings Warm	**5 Ball Bearings Cold**	**1/4" steel pin Warm**	**1/4" steel pin Cold**	**1/4" brass pin Warm**	**1/2" Ceramic Warm**	**1/2" Ceramic Cold**
Wire deflection angle	4.828	4.828	1.187	1.187	1.187	1.326	1.326
Lift Left side	130	100	550	500	430	300	520
	150	120	580	410	500	320	450
	90	80	600	550	380	325	550
	80	100	550	520	400	400	350
	90	100	550	620		450	680
	80	150		550		340	500
		120		450		320	550
				650		410	600
				560		410	480
						420	400
							480
Left average [g]	**103**	**110**	**566**	**534**	**428**	**370**	**505**
Left stdev	**29**	**22**	**23**	**76**	**53**	**54**	**91**
Lift Right side	140	150	600	560	520	350	450
	160	200	600	580	600	340	580
	150	150	580	550	500	350	450
	140	170	600	560	500	340	480
	120	200	620	620		500	550
		150	580	480		420	600
		120	600	550		420	480
		180		550		480	400
		150		500		540	600
		150		560		400	620
		180		680			580
		170		480			680
Right average [g]	**142**	**164**	**597**	**556**	**530**	**414**	**539**
Right stdev	**15**	**24**	**14**	**56**	**48**	**72**	**85**
Grand average [g]	123	137	582	545	479	392	522
Corrected for Tare [g]	98	112	557	520	454	367	497
Average Tension [g]	1193	1290	1290	1290	1290	1290	1290
Weight (=tension*angle)	5760	6231	1532	1531	1531	1711	1711
Friction coefficient	**0.017**	**0.018**	**0.363**	**0.340**	**0.296**	**0.214**	**0.291**

Results

Table 1 has all the data and results.
We see that the pulley system has a friction coefficient rho of 0.018.
The bearings were of the cheap, unground variety, and had large clearances.
This is actually an advantage here in that it minimizes friction.

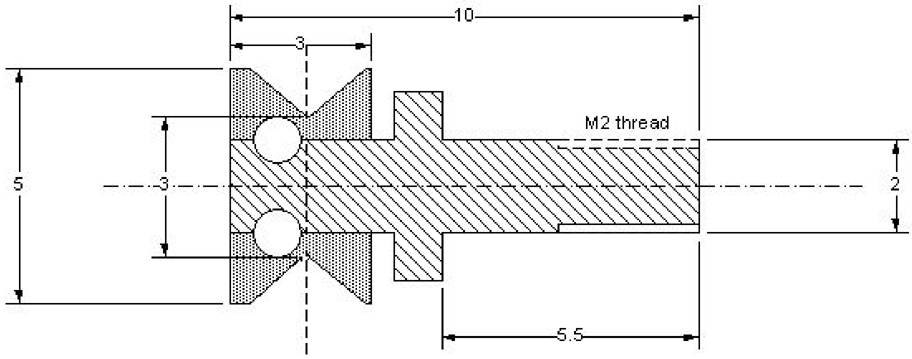
The steel pin had a static friction coefficient rho of 0.36 warm and 0.34 cold. The difference is not significant.
Brass had a static friction coefficient rho of 0.296 warm (it was not measured cold).
Glazed porcelain ceramic (in the form of a 1/2 inch diameter electronic standoff) had a coefficient of 0.21 warm and 0.29 cold. The coefficients may have been measured lower than actual due to brass contamination. Different parts of the wire seemed to behave differently.

Discussion

We can derive a value for the largest acceptable friction coefficient from the experiment requirements. A reasonable requirement would be that we do not have a high likelihood of two neighboring wires touching. For a wire spacing of 6 mm that means we want the expected sagitta uncertainty to be under 3 mm in most case, because the static friction can increase or decrease the wire tension, depending where the motion stops.
The friction coefficient rho must, hence, be smaller or equal to this tolerance divided by the wire sagitta.
For a 30 m long steel wire of 0.15 mm diameter, installed 30 degrees from the vertical, and tensioned with 1300 g, we expect a sagitta of 4 mm.
The friction coefficient must therefore be less than or equal to 3mm/4mm = 0.75
The requirement is met by the measured friction coefficients.
If the wire deflection is 30 degrees, then the contact force is half the wire tension, and the friction “coefficient” becomes 0.2
In practice, static friction is quite variable.
In cases where one pulls on one end of the wire, as opposed to reducing the tension on the other end, The wire may self-lock and not move at all.
However, even this dire situation is not fatal; the wire can accommodate the strain from cooling shrinkage while developing only moderate stress, about 10 ksi.
For now, my conclusion is that some friction relief is needed.
Pulleys of all types will work, and I am also looking into low friction plastic bearing materials.

Outlook

I have made a sketch of acceptable dimensions for a ball bearing pulley,
And sent it to a couple of large manufacturers for comment.
(It can also be used to design a sleeve bearing pulley, of course).
This is shown in Fig. 3 below.



Notes:

Units are in mm.
Dimensions have about +- 1 mm leeway.
This serves as a pulley for wire tensioning.
Slow turning, only a few fewtotal turns
Radial load is 2 # in service, less than 5# during installation.
Wire diammeter is 0.15 mm.
Will be used in liquid Argon, at 70 Kelvin
Acceptable materials include Aluminum ,Stainless steel,
some non-halogenated plastics (e.g. glass filled epoxy and Udel)
if able to withstand temperature cycles and show no creep.
Large clearances are OK, e.g. axle can be stainless steel, race aluminum.
Looking for a lowcost solution
Quantity above 50,000 pieces
Must be ultra reliable against catastrophic failure such as coming apart.
Need failure rate under one in 50 million.
We may be able to perfom the 100% testing.

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Conceptual Dimensions for Wire Pulley

Figure 3